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RESEARCH MEMORANDUM

INVESTIGATION OF FLOW FLUCTUATIONS AT THE EXIT OF A RADIAL-FLOW CENTRIFUGAL IMPELLER

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NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS
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INVESTIGATION OF FLOW FLUCTUATIONS AT THE EXIT OF A RADIAL-FLOW CENTRIFUGAL IMPELLER

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SUMMARY

Surveys were made at the exit of a radial-flow centrifugal impeller to obtain instantaneous values of velocity from blade to blade and at various positions between the front and the rear diffuser walls. Surveys were also made at several radial stations midway between the walls of the diffuser to observe the radial change in the flow pattern through the vaneless diffuser. All surveys were made with a constant-temperature hot-wire anemometer.

The surveys showed that there was a decrease in velocity across the passage exit from the driving face to the trailing face of the impeller blade. In general, this decrease was greater at the front diffuser wall than at the rear diffuser wall. At midpassage height, the decrease was approximately 8 percent of the average velocity at a weight flow of 16 pounds per second and approximately 16 percent at a weight flow of 32 pounds per second. There were variations in this general flow pattern from passage to passage, and for the same passage from revolution to revolution, with turbulent velocity fluctuations of the order of 3 to 5 percent imposed upon the general pattern. This general flow pattern persisted into the diffuser and was still evident at a radius 23 percent greater than that of the impeller.

INTRODUCTION

Experimental results show (references 1 to 3) that the loss distribution within a rotating impeller passage is not uniform and that the air particles of low kinetic energy tend to accumulate along the blade trailing face with the result that the relative exit velocity at the trailing face is lower than that at the driving face. The data in references 1 to 3 were obtained for flow at average blade height.

In an attempt to measure the variation in absolute velocity with respect to time at a number of points between the hub and the shroud, as well as at the average blade height, hub-to-shroud surveys were made at the NACA Lewis laboratory with a constant-temperature-type hot-wire anemometer at the impeller exit. Surveys were also made in the radial direction midway between the front and the rear walls of a vaneless diffuser in order to observe the change in the flow picture during diffusion.

The results presented herein are qualitative, and interpretation of the data was made in light of the results obtained with the internal instrumentation described in references 1 and 3. Although the results are only qualitative, they do show the problems that may be encountered in diffuser design and give increased insight into the complex nature of the flow leaving a centrifugal impeller.

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APPARATUS AND PROCEDURE

A cutaway view of the 48-inch-diameter impeller test rig and a cross-sectional view showing the hot-wire probe locations are presented in figure 1. Except for the hot-wire anemometer setup, the apparatus and instrumentation are identical with that of reference 1.

A 0.0002-inch-diameter tungsten wire 0.040 inch long with a constant-temperature-type hot-wire-anemometer amplifier (references 4 and 5) was used to obtain the surveys. An oscilloscope with a triggered sweep was used to observe the variation in flow configuration. The sweep could be adjusted to give the exit flow configuration for successive revolutions of the impeller, for a given group of successive impeller passages. Four of the total of eighteen passages were chosen. The traces were recorded on 35-millimeter film which was moved past the oscilloscope screen at a constant rate. The excitation required to trigger the sweep was furnished by a permanent magnet that was attached to the impeller shaft and passed a stationary coil. Hub-to-shroud surveys from the rear diffuser wall to the front diffuser wall were made 1/2 inch from the exit blade tip with the wire perpendicular to the measured average flow direction at each survey position and lying in a plane parallel to the rear diffuser wall. Surveys were also made at the exit blade tip from $\frac{1}{8}$ to $5\frac{1}{2}$ inches radially outward midway between the diffuser walls with the wire parallel to the axis of rotation. The surveys were taken at equivalent weight flows $W\sqrt{\theta}/\delta$ of 16, 26, and 32 pounds per second at an equivalent tip speed $U/\sqrt{\theta}$ of 700 feet per second. (All symbols are defined in the appendix.) The performance over a range of weight flows at this tip speed is shown in figure 2.

Because of diffuculties encountered in obtaining quantitative data with the hot-wire anemometer in this application, quantitative data are not given except as follows: The velocities closest to the driving and trailing faces of the blades near the exit as measured internally (reference 1) were used to determine the approximate magnitude of the absolute velocity in the diffuser. It was assumed that the flow was parallel to the blades in this vicinity with zero slip. These values were used to give an approximate calibration for the traces at the midpassage height for each survey. Inasmuch as little or no variation in density across the passage occurred near the exit for the internal measurements, the variation shown by the hot-wire anemometer were considered to be due to velocity variation only.

2658 Each survey was made without replacing the probe or changing the instrument settings. Therefore, for any given figure in which the results of a survey are shown, the mass-flow scale represented by the amplitude is constant over the survey range except for Mach number effects.

In order to prevent excessive noise signals due to 60 cycle pickup and high-frequency sinusoidal signals which were probably caused by probe and wire vibration from appearing on the oscilloscope, a band-pass filter having a lower cut-off frequency of 60 cycles per second and an upper cut-off frequency of 7000 cycles per second was used. The blade wake frequency was 1000 cycles per second.

RESULTS AND DISCUSSION

Typical Trace

Orientation. - A typical set of traces as photographed on the oscilloscope screen is shown on figure 3. Each trace shows the velocity profiles at the exits of four complete passages with the decrease in velocity going from the driving face of one blade across the passage to the trailing face of the preceding blade. The straight, almost vertical, portions of the traces show the jump in velocity from the trailing face to the driving face of the blade. The three traces represent the same four passages for succeeding revolutions of the impeller. The over-all slope of each trace is the result of a combination of film movement and sweep velocity. The slight variation in slope is due to the uneven sweep velocity of the oscilloscope. The base slope line was obtained for the succeeding figures by attenuating the amplitude and photographing the sweep.

Flow variation. - These traces indicate that variations in the flow patterns may be expected from passage to passage and for a given passage from revolution-to-revolution. Further evidence of the variations from passage to passage is given in figures 4 to 6. There are roughly three components of variation; a general decrease in velocity across the passage from blade to blade because of uneven loss distribution within the impeller, a V-shaped wake shown by the straight, almost vertical lines at the blade exit, and turbulent fluctuations imposed upon the general pattern. In subsequent discussions, any reference to decrease in velocity across the passage from blade to blade will be for the general decrease and not for that in the blade wake. The magnitude of the variation in velocity from maximum to minimum in figure 3 is of the order of 45 feet per second, which is about 7 percent of the absolute velocity.

From comparison of the velocity profiles with the internal data, it appears that the minimum point of the wake represents a velocity very nearly equal to the impeller tip speed. This drop in air velocity in the wake to that of the impeller tip at the exit can be more easily observed on traces recorded at the higher weight flows in which the wake is more nearly symmetrical about the blade tip (fig. 5(b), for example). For low weight flows, the drop in velocity between blades to a value below that of the impeller tip is due to a combination of low relative velocity and slip factor (fig. 4(f), for example). 2658

The tangential velocity is lower than the tip speed by virtue of the fact that the slip factor is less than 1. At low weight flows, where the radial component is low, the absolute component drops below the wheel speed.

Hub-Shroud Surveys

General observations. - The change in the velocity-profile shape from blade to blade from the rear diffuser wall to the front diffuser wall is shown in figures 4 to 6 for equivalent weight flows of 16, 26, and 32 pounds per second, respectively.

At midpassage height $(D = 0.5)$, the decrease in velocity from the driving face to trailing face across the passage increases from approximately 8 percent of the average velocity at a weight flow of 16 pounds per second to approximately 16 percent at a weight flow of 32 pounds per second. Except for the 16-pound-per-second weight flow, the decrease became more severe in going from the rear to the front diffuser wall; the flow became extremely turbulent in the vicinity of the front diffuser wall for all weight flows. That sizeable losses are associated with the flow along the stationary shroud and the front diffuser wall has been indicated by total-pressure surveys at the exit of several experimental impellers as well as at the exit of the impeller used in this investigation. These losses may be caused by boundary-layer separation and clearance losses. The low-energy air produced by these losses may in turn be shifted to the trailing-face side of the passage through secondary-flow effects.

Boundary layer. - With the large decelerations which occur in impellers designed for considerable diffusion within the impeller passage, boundary-layer build-up may become a major source of loss, especially along the trailing face of the blade. The deceleration rate along the

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trailing face near the impeller entrance increases with decreasing weight flow in this compressor as shown by the internal data of reference 3 and may have been responsible for large boundary-layer losses at a weight flow of 16 pounds per second. For air weight flows of 26 and 32 pounds per second, the decrease in velocity across the passage, which is caused by internal losses along the trailing face of the blade as shown by the traces, is relatively slight at $D = 0.07$ and increases at larger values of D. However, at a weight flow of 16 pounds per second the decrease is evident across the entire passage height.

Clearance losses. - Although the velocity usually drops off from the driving to the trailing face across the passage at the exit, it is generally lower on the driving face of the blade than on the trailing face because of blade loading in regions other than near the exit. The higher pressure on the driving face causes spillage of this low-velocity high-pressure air through the clearance space to the trailing-face side into the region of lower static pressure. The introduction of the lower-velocity particles reduces the velocity in the region of the trailing face. An increase in velocity of the particles from the driving face may be expected because of the expansion of the air into a region of lower pressure. However, the increase is in the direction of the expansion which is generally transverse to the through-flow direction. As a result, the increase in velocity serves mainly to increase the mixing losses. Reduction of the trailing-face velocity in this manner would produce the greater decrease near the shroud as shown in figures 4 to 6.

Secondary flows. - Movement of low-velocity air to the trailing face may also result from secondary circulation. A possible explanation is as follows: In figure 7(a) is shown a particle on a radial streamline with the forces which act when there is no movement normal to the streamline; if the velocity should be reduced with the pressure gradient dp/dn remaining constant, the particle would move toward the trailing-face side of the passage. It was found experimentally (reference 1) that there was little or no variation in static pressure from hub to shroud within the impeller for a given radial position. Hub-to-shroud velocity surveys at the diffuser entrance showed a pattern similar to that of figure 7(b). If this velocity pattern exists within the impeller passage with no variation in static pressure from hub to shroud, the fluid will move in the direction shown in figure 7(c). With the movement of low-velocity particles from the driving to the trailing face along the shroud, the decrease in velocity shown by figures 4 to 6 would result.

Velocity fluctuations. - The traces presented in figures 4 to 6 were selected for each survey position from a large number of traces at that position. The traces selected were considered typical in that the velocity decrease from driving-to-trailing face in each case was very near the average and the imposed fluctuations followed no general pattern. These imposed fluctuations show the highly turbulent nature of the flow at the exit. The extremely large fluctuations near the front diffuser wall probably were caused by the blade-to-shroud clearance. Except in the vicinity of the front diffuser wall, the fluctuations were of the order of 3 to 5 percent of the average velocity.

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Rapid variations in over-all flow which might occur as the passage passes the hot wire cannot be distinguished from those due to turbulent fluctuations. However, hot-wire surveys just upstream of the impeller entrance where turbulence was negligible showed no rapid variation in over-all flow except at the surge weight flow of 14 pounds per second.

Radial Surveys

General observations. - There is generally a substantial decrease in measured total pressure from the immediate exit of an impeller to a distance downstream in the diffuser approximately one-half the impeller radius (reference 6). This drop in total pressure is probably due to the mixing losses which arise from the large decrease in velocity across the blade tip from the driving to the trailing face of the blade and the imposed fluctuations. In addition, a total-pressure probe gives a higher than average total pressure in a fluctuating stream which would account, in part, for a higher indicated pressure near the impeller exit.

Radial-flow changes. - Surveys were made in the radial direction to observe the change in the flow picture during diffusion. The results of the surveys are shown in figures 8 to 10. The distance from the impeller tip is given in terms of R, the ratio of the radius at the survey point to the impeller-exit radius. The variations persist over the entire survey range for all weight flows. At the final survey station ($R = 1.23$), the mass-flow variations diminished to around 75, 40, and 60 percent of the original value at weight flows of 16, 26, and 32 pounds per second, respectively. The low value for the 26-pound-per-second weight flow is in agreement with the higher measured efficiency shown for this weight flow in figure 2.

Diffuser problem. - The mixing losses caused by the impeller take place in the diffuser. The evaluation of a diffuser as part of a composite compressor unit is therefore very difficult. On the other hand, evaluation as a separate component may be misleading. For example, if the flow leaving an impeller follows a corkscrew path because of secondary flow, it may be possible to design a vaned diffuser in such a manner as to eliminate the corkscrew motion by inducing secondary flow in the opposite direction. As another example, a vaneless diffuser designed to turn the flow in the meridional plane may be highly efficient as a separate component, whereas the variation in velocity in the tangential direction may cause prohibitive secondary flow losses when the diffuser is placed at the exit of an impeller. With these conditions, it may be desirable to investigate diffusers of varying design by evaluating their relative performances with a given impeller.

SUMMARY OF RESULTS

Surveys were made at the exit of a radial-flow centrifugal impeller to obtain instantaneous values of velocity from blade to blade and hub to shroud. Surveys were also made in the radial direction midway between the walls of the diffuser to observe the change in the flow pattern during diffusion. All surveys were made with a constant-temperature hot-wire anemometer, and the following results were obtained:

1. There was a decrease in velocity across the passage exit from the driving face to the trailing face of the blade. In general, this decrease was greater at the front diffuser wall than at the rear diffuser wall. At midpassage height, the decrease was about 8 percent of the average velocity at a weight flow of 16 pounds per second and approximately 16 percent at a weight flow of 32 pounds per second.

2. Turbulent velocity fluctuations of the order of 3 to 5 percent were imposed upon the general through flow.

3. There were variations in the flow patterns from passage to passage, and for the same passage from revolution to revolution.

4. Very large and incoherent fluctuations in velocity occurred along the front diffuser wall; these were probably caused by the blade-to-shroud clearance.

5. The general flow pattern was still evident at the final radial-survey station which was at a radius 23 percent greater than that of the impeller; the mass-flow variations had diminished to around 75, 40, and 60 percent of the original value at weight flows of 16, 26, and 32 pounds per second, respectively.

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APPENDIX - SYMBOLS

The following symbols are used in this report:

D ratio of d to d_t

d distance from rear diffuser wall (fig. 1(b)) 2658

d_t total distance from front diffuser wall to rear diffuser wall at survey station (fig. 1(b))

n distance along normal to streamline (fig. 7)

p static pressure

q velocity relative to impeller

R ratio of radius at survey point to impeller-exit radius

r radius at survey point, (fig. 1(b))

U impeller-tip speed, ft/sec

W total compressor weight flow, lb/sec

δ ratio of total pressure at entrance to standard sea-level pressure

θ ratio of inlet stagnation temperature to standard sea-level temperature

ρ mass density, $lb/(sec^2)(ft^4)$

ω angular velocity of impeller, radians/sec

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5. Lowell, Herman H.: Design and Application of Hot-Wire Anemometers for Steady-State Measurements at Transonic and Supersonic Airspeeds. NACA TN 2117, 1950.

6. Ginsburg, Ambrose, Johnsen, Irving A., and Redlitz, Alfred C.: Determination of Centrifugal Compressor Performance on Basis of Static-Pressure Measurements in Vaneless Diffuser. NACA TN 1880, 1949.

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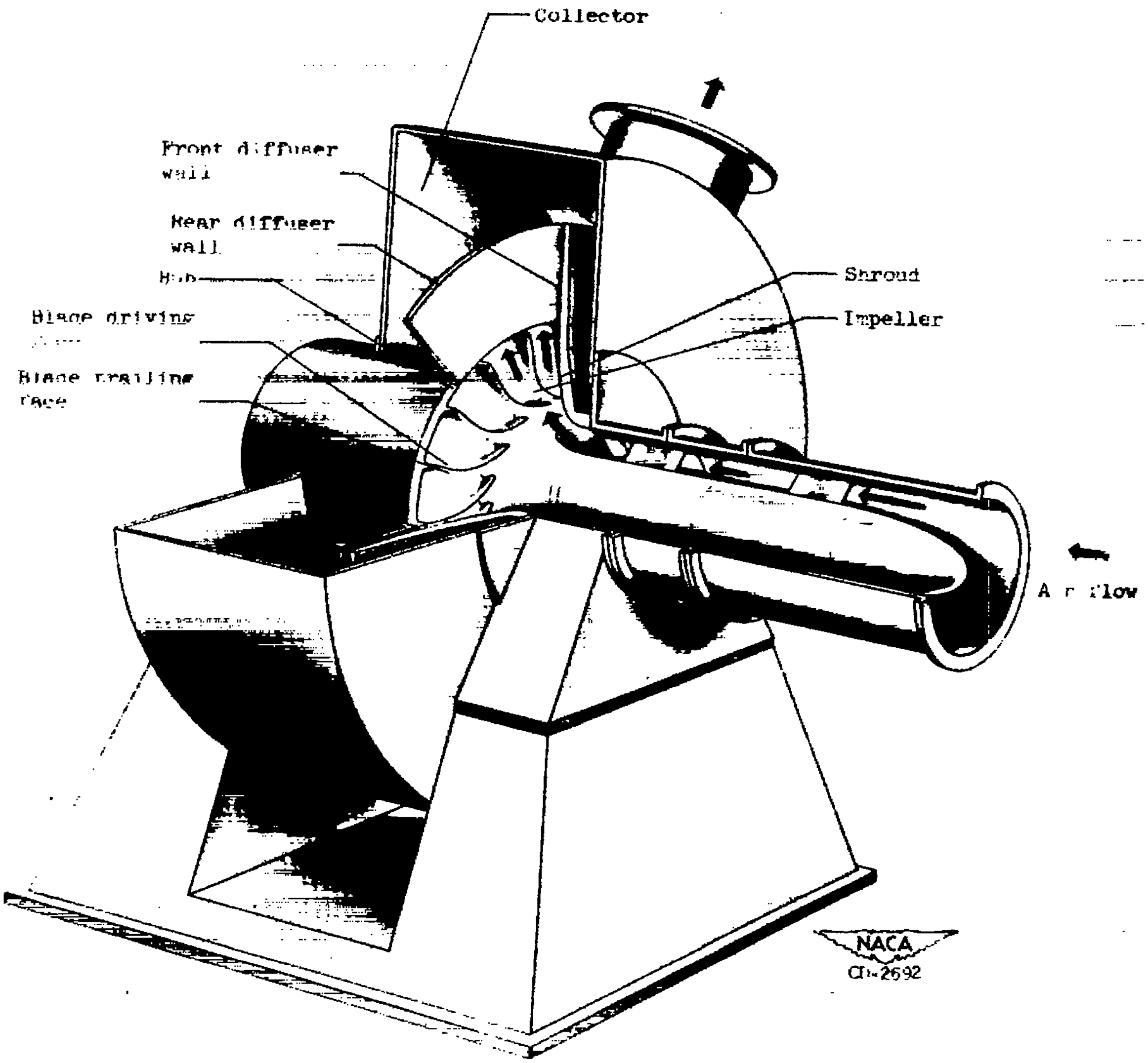

(a) Cutaway view.

Figure 1. - 48-Inch impeller test rig.

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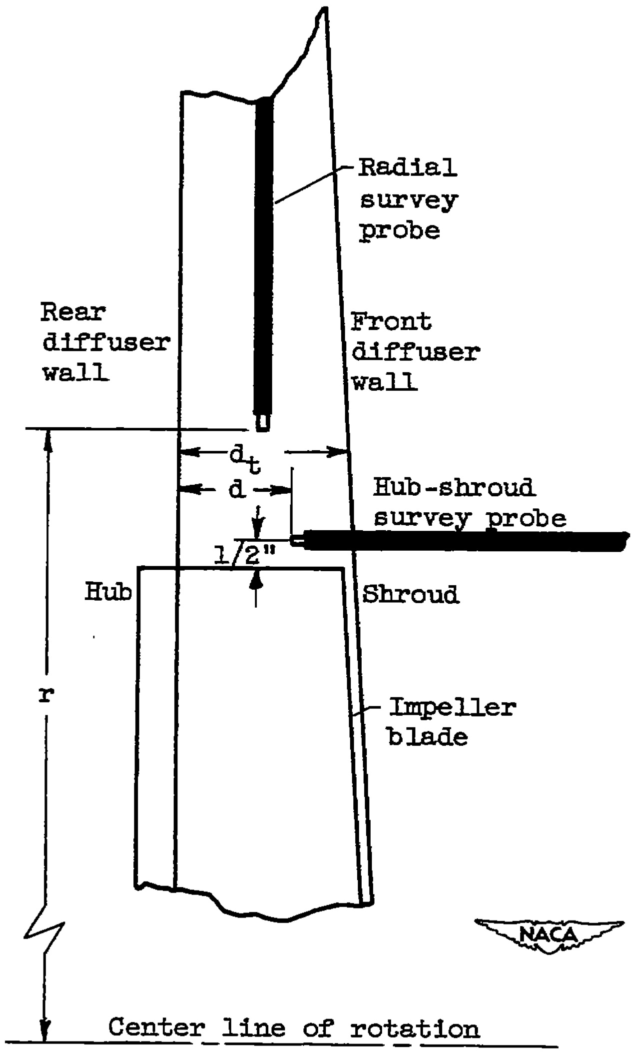

(b) Hot-wire-probe locations.

Figure 1. - Concluded. 48-Inch impeller test rig.

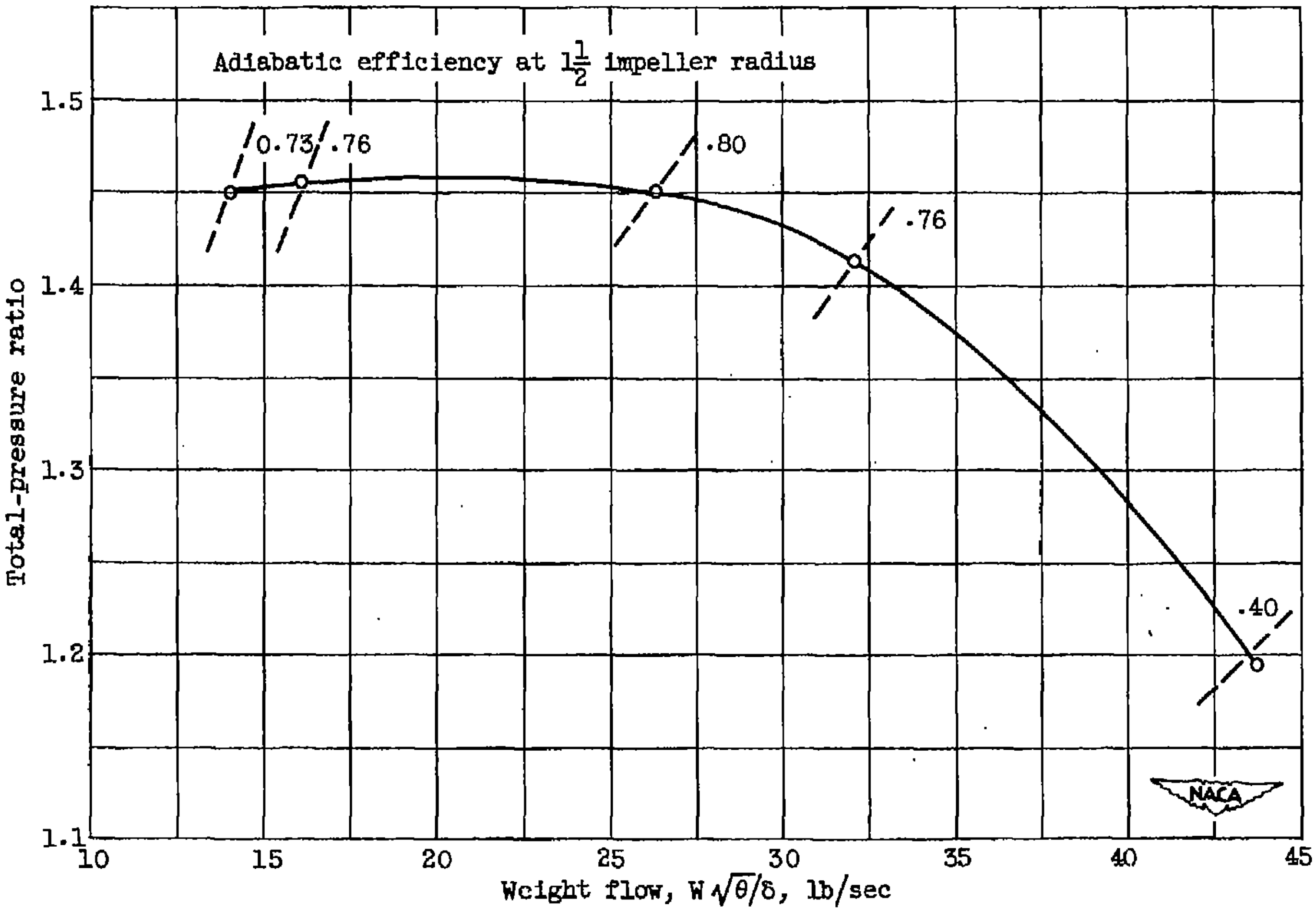

Figure 2. - Over-all performance at equivalent tip speed of 700 feet per second.

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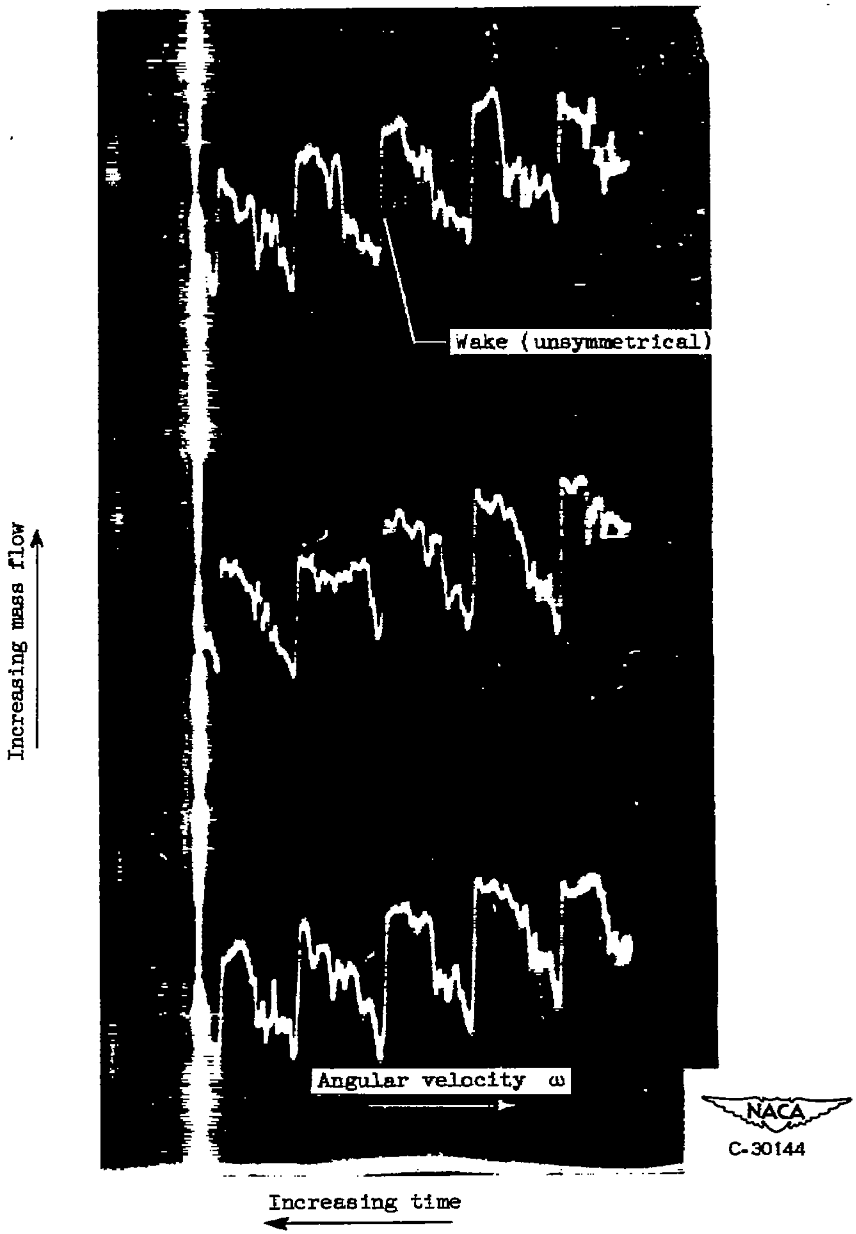
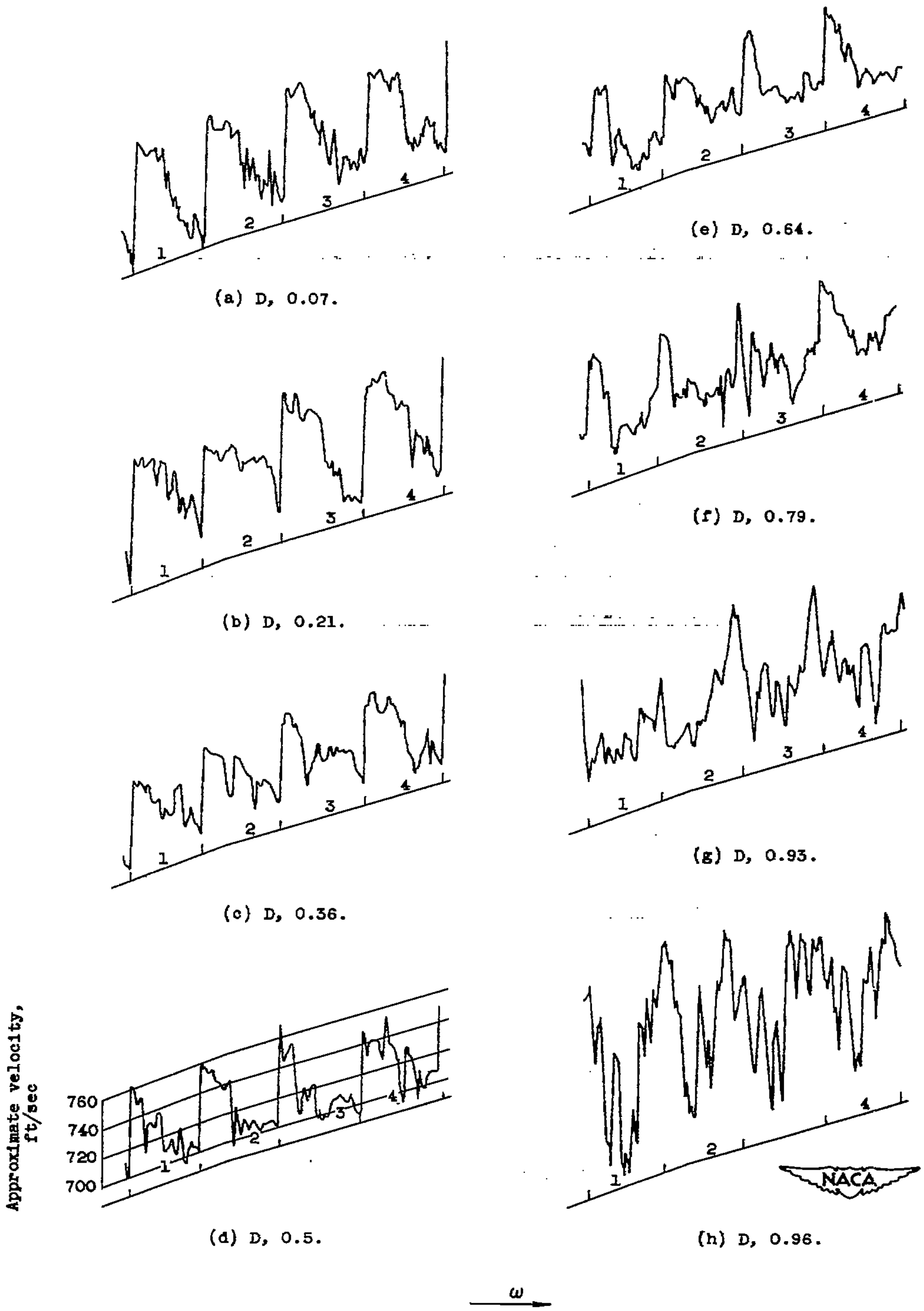


Figure 3. - Typical set of traces as photographed on oscilloscope.



(a) D, 0.07.
(b) D, 0.21.
(c) D, 0.36.
(d) D, 0.5.
(e) D, 0.64.
(f) D, 0.79.
(g) D, 0.93.
(h) D, 0.96.

Figure 4. - Hub-to-shroud survey with velocity variation across blade passage at exit (four passages). Equivalent weight flow, 16 pounds per second. Driving-to-trailing face across passage is in direction of rotation ω. (Numbers refer to successive passges.)

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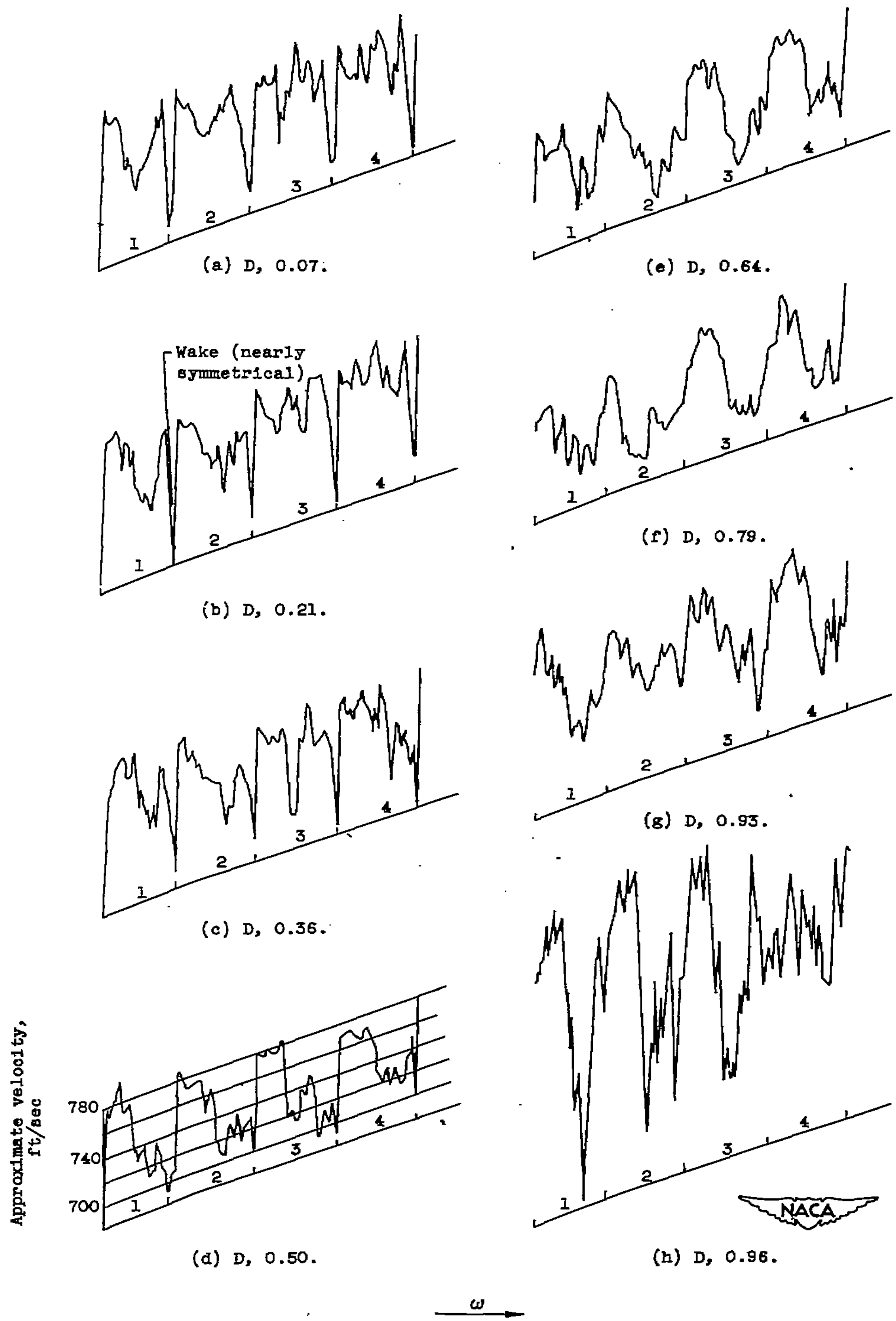


Figure 5. - Hub-to-shroud survey with velocity variation across blade passage at exit (four passages). Equivalent weight flow, 26 pounds per second. Driving-to-trailing face across passage is in direction of rotation ω. (Numbers refer to successive passages.)

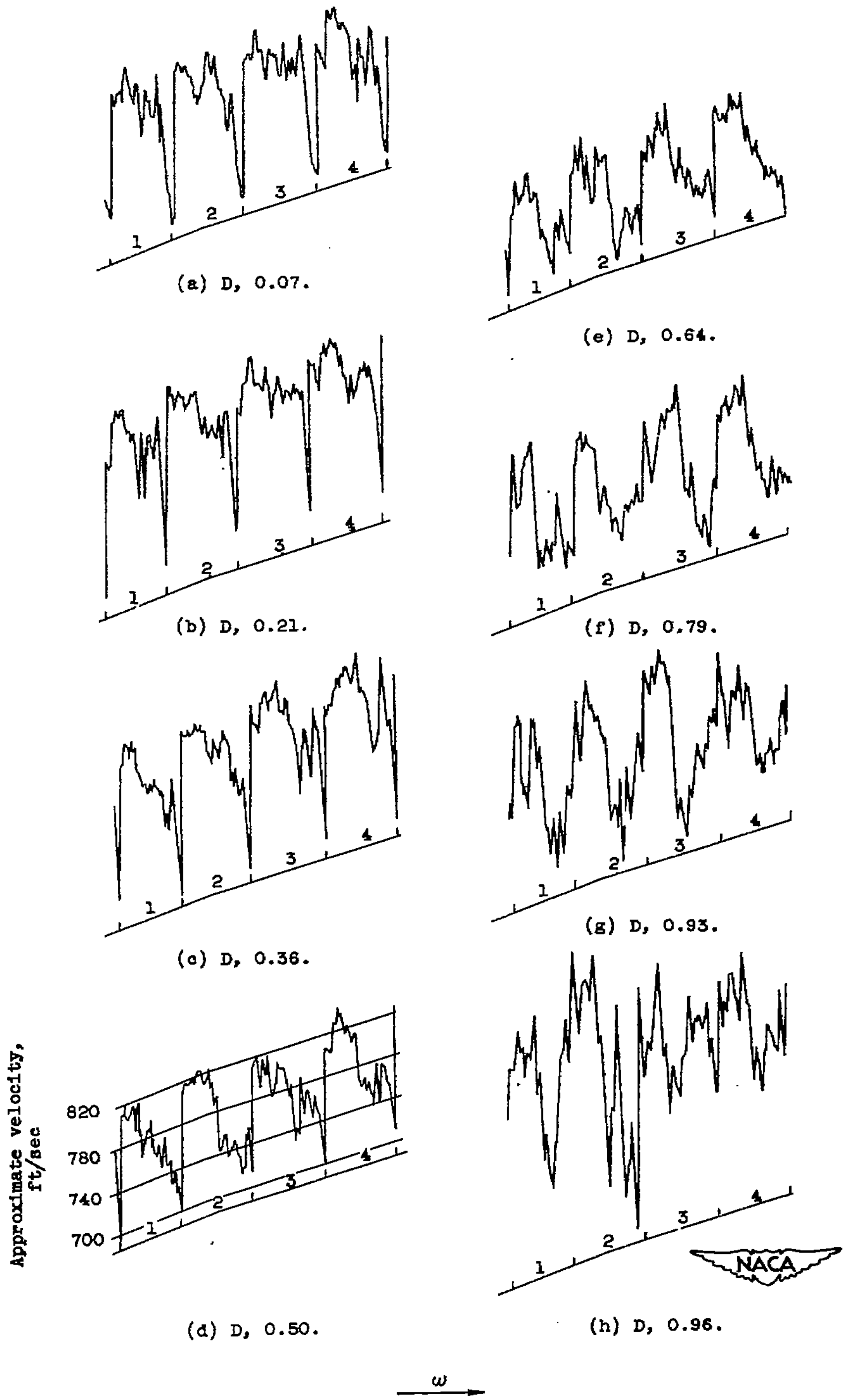
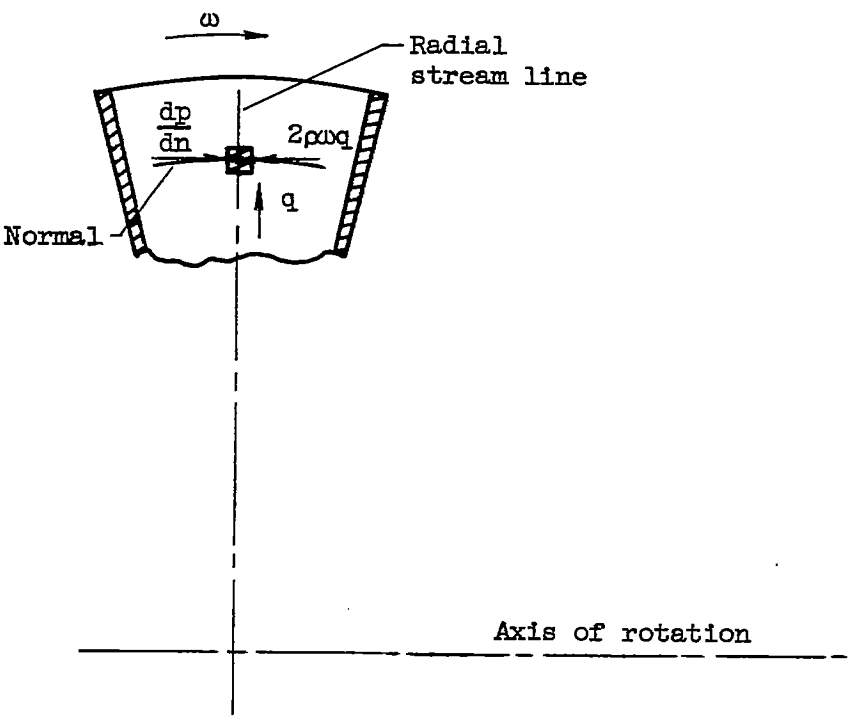
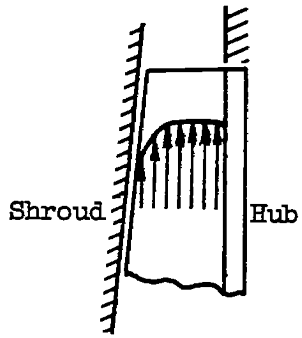


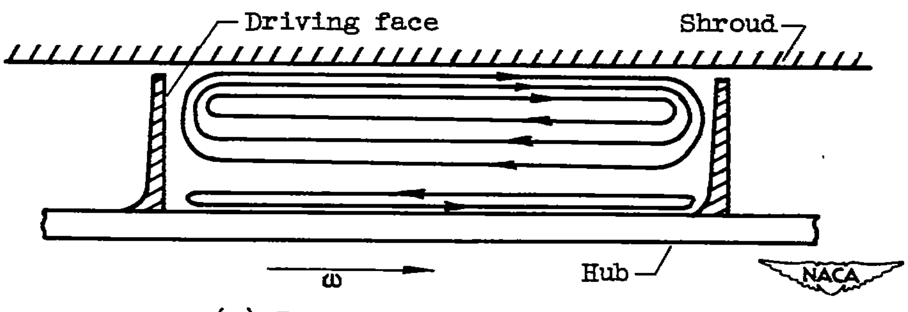
Figure 6. - Hub-to-shroud survey with velocity variation across blade passage at exit (four passages). Equivalent weight flow, 32 pounds per second. Driving-to-trailing face across passage is in direction of rotation ω. (Numbers refer to successive passages.)



(a) Impeller passage.



(b) Hub-shroud profile.



(c) Passage cross section.

Figure 7. - Secondary flows in rotating passage.

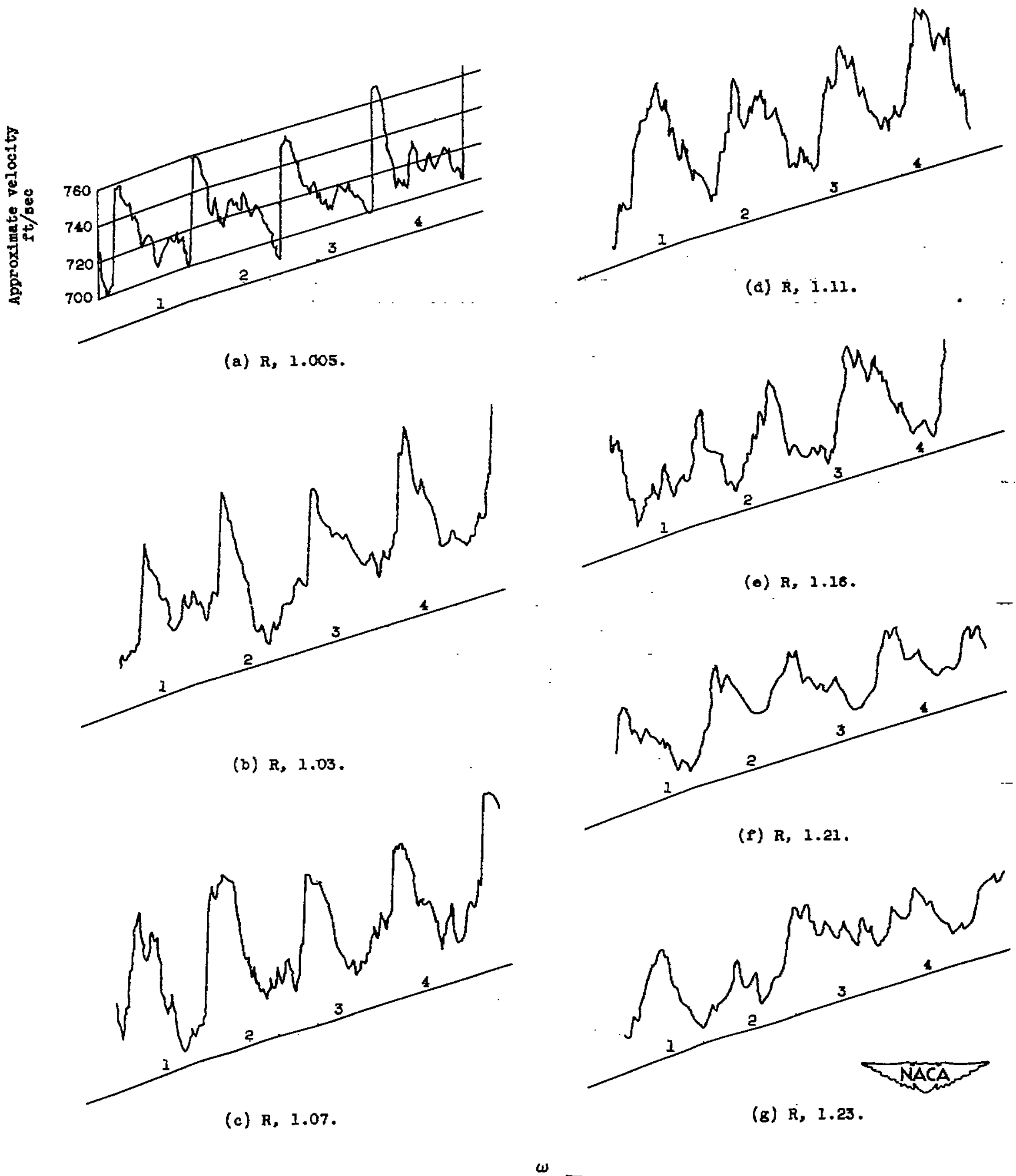


Figure 8. - Radial survey in diffuser with velocity variation across four passages. Equivalent weight flow, 16 pounds per second. Driving-to-trailing face across passage is in direction of rotation ω. (Numbers refer to successive passages.)

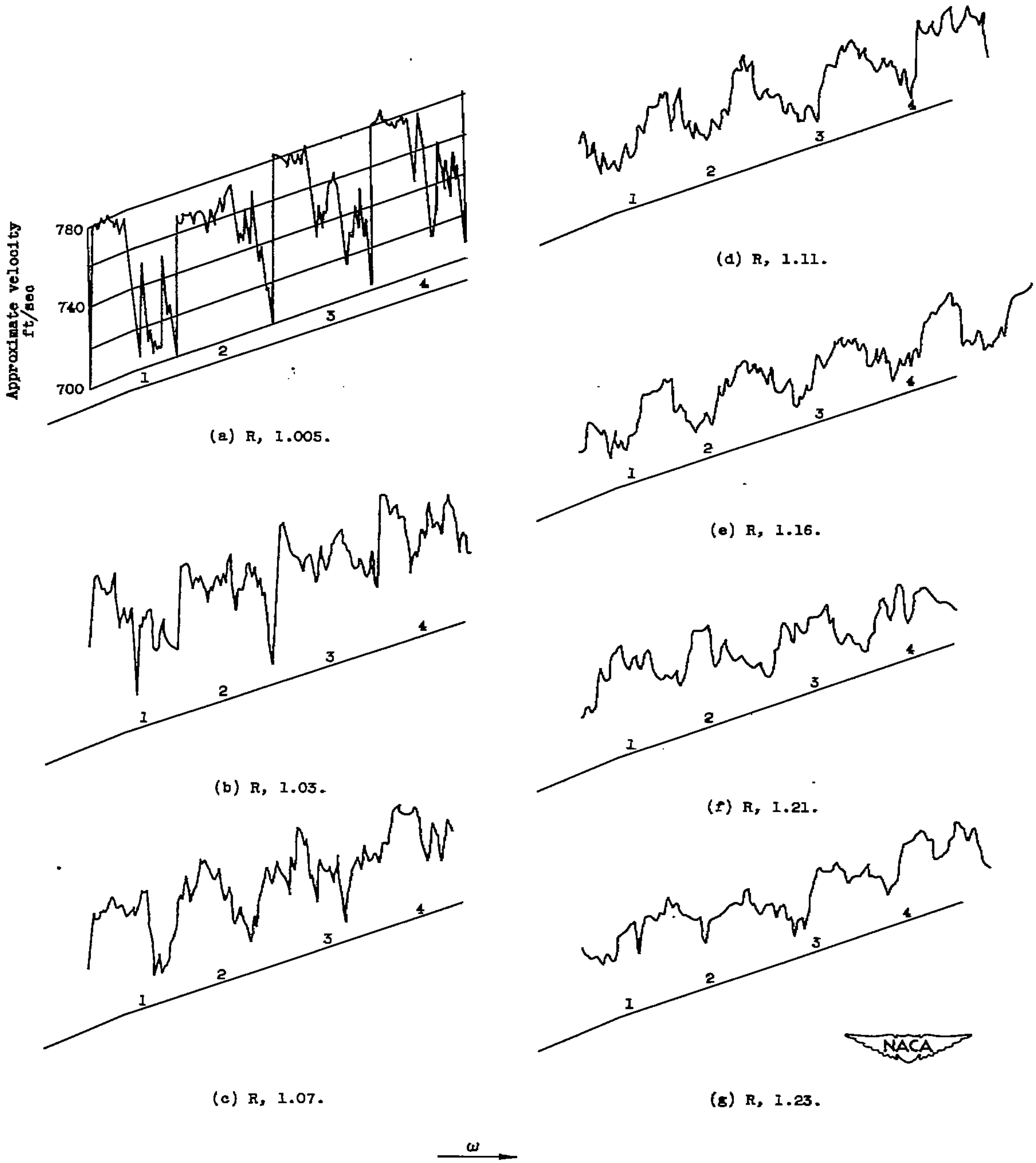


Figure 9. - Radial survey in diffuser with velocity variation across four passages. Equivalent weight flow, 26 pounds per second. Driving-to-trailing face across passage is in direction of rotation ω. (Numbers refer to successive passages.)

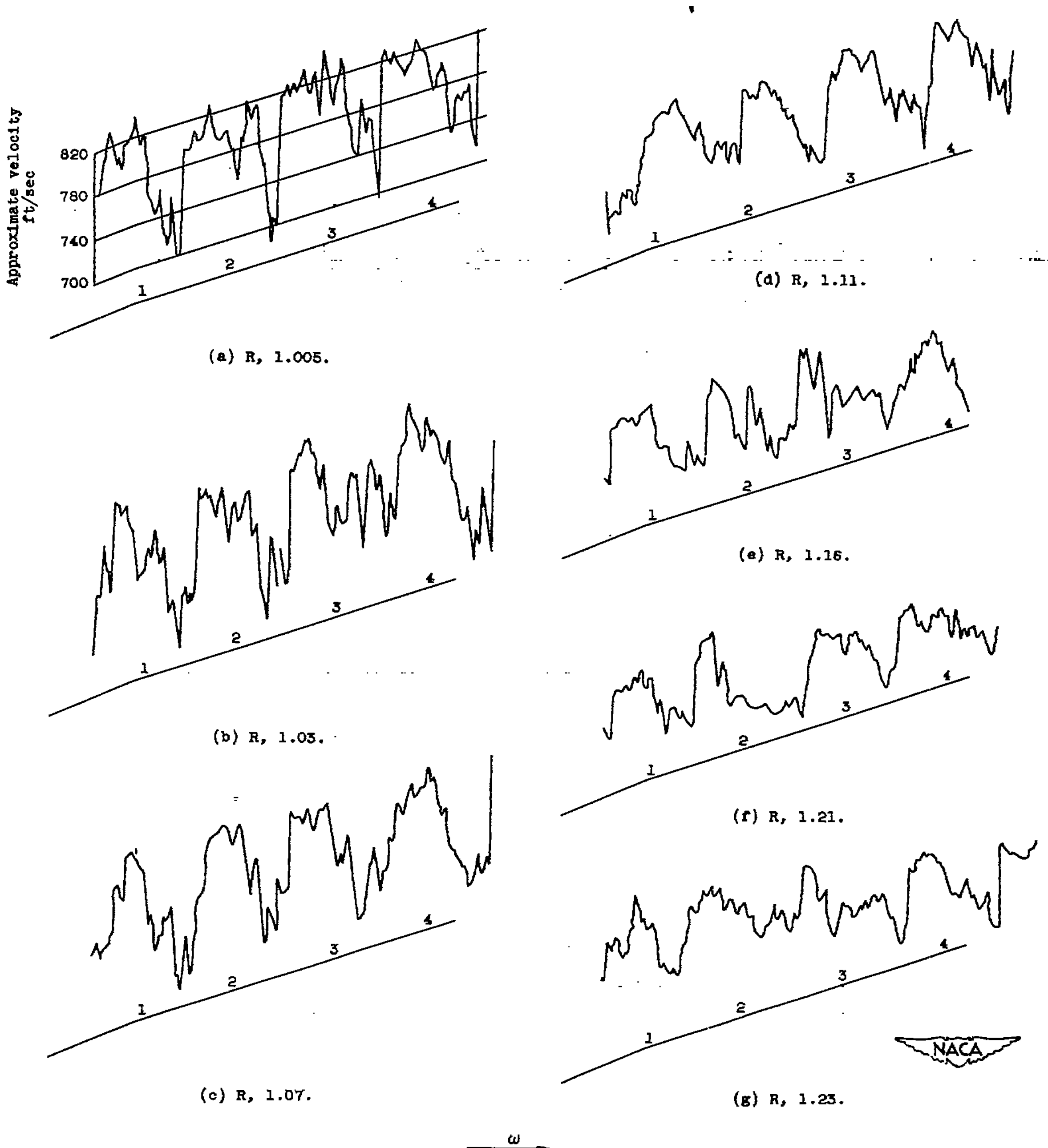


Figure 10. - Radial survey in diffuser with velocity variation across four passages. Equivalent weight flow, 32 pounds per second. Driving-to-trailing face across passage is in direction of rotation ω. (Numbers refer to successive passages.)